Seasonal Performance Evaluation Procedures for Domestic Electric Driven Heat Pumps and Air Conditioners

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INTRODUCTION

Virtually[1] all of the residential and commercial heat pumps available in today's market operate on the same fundamental principles as do, that of the vapor compression cycle. Also, virtually all of these systems use fluorocarbon compounds as a working fluid and have a constant speed positive displacement compressor with leaf spring valves[2]. This concept has been continuously developed throughout this century and has served society well; however, it contains a couple of inherent limitations which cause any particular design to perform somewhat less than ideally. The first and most significant of these limitations is due to the fact that a refrigerant's density is proportional to its pressure. Therefore, when the evaporator temperature is required to be decreased, the saturation pressure must also be decreased with a corresponding reduction in the circulating refrigerant suction density [positions (1 through 5) of Fig. 1], causing a loss in system capacity. This capacity loss is accented by the use of constant speed piston compressors whose valves operate on a differential pressure between the interior cylinder conditions and either the suction or discharge line pressures. If either the condenser or evaporator temperature conditions increase or decrease, respectively, less valve open time will exist per stroke and thus less refrigerant will be pumped; leading to an additional loss in capacity. This capacity reduction is particularly significant in the heat pump heating application for residences since it occurs simultaneously with an increase in building transmission (and thus heat pump) load.

Of course, this overall decrease in refrigerant mass flow rate which causes the decrease in capacity also causes a decrease in compressor work. However, since the pressure difference or lift has increased the work per unit mass of refrigerant has increased and thus the compressor work never decreases as much as the capacity, resulting in a net decrease in heat pump efficiency or coefficient of performance (COP). This steady state part load phenomena has the most significant influence on the seasonal performance of today's heat pumps and thus must be determined most carefully in any evaluation procedure.

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[1]A very small percentage of the air conditioning in the U.S. is based on the thermally activated absorption cycle.

[2]Screw, rotary and scroll compressors use a different valving system.

Heat pump operation in the heating mode is unique among the family of refrigeration machines in that the desired commodity is the condenser output instead of the evaporator input. Residential heat pump applications in the U.S. require the same machine to do both heating and cooling, thus the most important parameter, capacity, alternates between condenser output for heating and evaporator input for cooling and dehumidifying. The machinery elements basic to the current dual mode heat pump designs are shown in Fig. 1. In addition to the four elements included in all refrigeration systems (i.e. condenser, evaporator, compressor, expansion device), a reversing valve and often an accumulator are required in a heat pump. Although the latter two have only secondary thermodynamic importance, the reversing valve allows the heat exchangers to be alternately used for condensing and evaporating purposes (which necessarily compromises their steady state performance). The accumulator's primary function is to protect the compressor from receiving a slug of liquid refrigerant particularly during the reversing periods demanded by the defrost system. This accumulator, also acting as a storage reservoir for that refrigerant which is not being circulated, tends to cause the system to respond slowly to changes in demand; particularly during start-up [1].

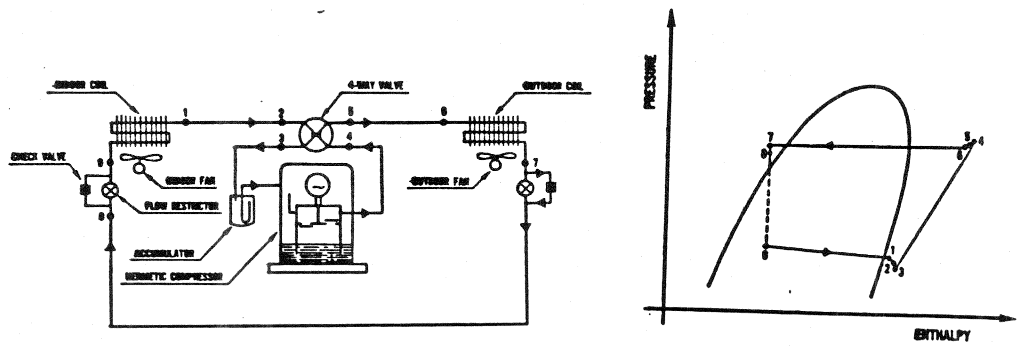

Fig. 1. Elements of a traditional heat pump

Similarly heat pump designs which incorporate oversized compressor crankcases, in lieu of accumulators can suffer from this same dynamic response degradation. Thus cycling losses are typically more significant for heat pumps than air conditioners. There can be either water or air media acting as either a source or sink fluid; however, the overwhelming majority of U.S. residential applications use air for both source and sink. This means that for the period in which the outdoor coil is acting as an evaporator (e.g. winter) frost will form degrading the system's capacity by blocking the air passages between the finned tubes. This frosting effect as well as the energy required to defrost the coil is an additional loss associated with today's current heat pump designs.

Although there are many other factors that will affect the heat pump's field performance (e.g. sizing, duct losses, etc.) it is primarily supplemental resistance heating and these two, cycling and frosting, that prevent the seasonal efficiency from being a simple weighted summation of steady state efficiencies at various outdoor temperatures. These factors along with sufficient steady state full and part load operational data are used to form the basis for today's air-to-air heat pump performance testing procedures.

Steady state efficiency is presented in a variety of forms, all of which are based on the same principle of the ratio of the desired commodity to the cost of the input. For steady state efficiency, the heating mode term is coefficient of performance (COP) defined as the condenser output in watt-hours to the energy consumed in watt-hours; where the heating output is enthalpy increase of the indoor air stream and the energy consumed is electric input to compressor, outdoor fan, indoor blower, crankcase heater and controls but <u>not</u> the supplemental resistance heat. The cooling mode efficiency term is the energy efficiency ratio (EER) defined as the enthalpy decrease of the indoor air stream (i.e. both sensible and latent heats) to the energy consumed which is again the total electric input. In principle the EER is identical to the COP except for the conversion factor of 3.413 Btu/watt hour. This somewhat non-rigorous scientific term, EER, has been agreed upon by the industry because the cooling efficiency is the primary parameter used by the energy conscious consumer for selection among units. It is felt that the wider spread between the EER rating number would be more easily distinguished between by the layman. For example, three units having COP values of 2.46, 2.70. 2.93 will have EER values of 8.4, 9.2, 10.0 respectively.

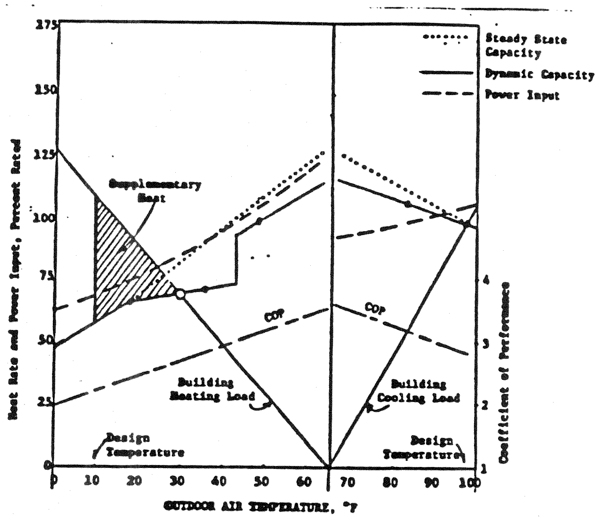

Fig. 2. Heating and cooling performance characteristics of a typical air-source heat pump.

Of course these steady state efficiency terms cannot recognize the cycling and frosting degradations that inevitably occur during field operation. For this purpose the heating seasonal performance factor (HSPF) and the seasonal energy efficiency ratio (SEER) were created. They are defined as total seasonal heating capacity or total seasonal cooling and dehumidifying capacity to total seasonal electric consumption in watt-hours, respectively.

Note, again the dimensioned efficiency format is used. In principle they are the same thing and are based on a weighted average (with number of hours in each 5°F temperature "bin") of the COP obtained by dividing the respective capacities and power values shown in Fig. 2. That is:

$$\frac{1}{SPF} = \frac{\sum_{j=1}^{k} n_j \, Load(T_j)[1/COP(T_j)]}{\sum_{j=1}^{k} n_j \, Load(T_j)} \qquad \text{where } SPF = \frac{\text{seasonal output}}{\text{seasonal input}} \qquad (1)$$

and where n_j is the number of hours within the 5°F temperature bin, j, for the region the unit is being rated, Load(T_j) is the respective load or capacity for the same bin and COP(T_j) is determined from the ratio of the dynamic capacity (solid line) to the power in the same temperature bin. This will account for the cycling and defrosting effects as prescribed by the rating criteria. The above equation applies in principle to either the heating or cooling season.

TEST AND RATING METHODOLOGIES

Because of its predominance in the US market the air to air heat pump is the unit exemplified in this paper to demonstrate how field performance may be characterized but not simulated or duplicated in a laboratory based rating scheme. Its performance is plotted in Fig. 2 where it is assumed that the unit is sized exactly for the cooling load design condition (the official DoE/NBS rating procedure assume 10% oversizing). The capacity decreases, with increasing saturation temperature differences between the evaporator and the condenser (i.e. as indoor and outdoor temperature differences grow), as indicated by the dotted curves.

The testing procedure for heat pumps in the cooling mode requires one certification point taken under full-load steady-state conditions at 35°C DB (95°F) outdoor temperature and 26.7°C DB (80°F)/19.4°C WB (67°F) indoor conditions. In order to establish a measure of the part-load cyclic effect it is necessary to evaluate the unit at one more outdoor temperature condition. The outdoor temperature at which this test is run is somewhat arbitrary; a value of 27.8°C (82°F) was selected, because it is the weighted mean of the U.S.A. national summer dry-bulb temperatures.[3] As in the existing full-load test, the

[3] Weighted in proportion to unitary air conditioner sales around the country.

indoor coil should be wet (same indoor conditions as the 35°C test) during the test, since dehumidifying is inseparable in the cooling mode. Unfortunately, cycling a unit under wet-coil conditions presents both accuracy and repeatability problems. Wet-bulb instrumentation systems have long time constants so that both room control and indoor air stream enthalpy measurements are unreliable under periodic conditions. It was, however, noted through a series of tests, run with the meticulousness that only a research lab could afford, that:

$$\left.\frac{COP_{CYC}}{COP_{SS}}\right|_{\text{dry coil}} = \left.\frac{COP_{CYC}}{COP_{SS}}\right|_{\text{wet coil}}$$

With the assumption that the above relationship is true in general, one can then deduce the part-load wet-coil performance from a set of two steady-state tests (one wet, one dry) and a cyclic dry-coil test, all at the same outdoor temperature condition. Based on typical thermostat designs which control units to cycle at approximately 3 CPH at 50% on-time, the recommended cyclic test operation is 2 CPH at 20% on-time, which corresponds to 6 minutes on/24 minutes off.

With this cyclic wet-coil value calculated by means of the previous equation, a performance line may be defined by it and the steady-state data point at 35°C (95°F) (See Fig. 2). A load weighing process similar to the traditional bin method may be conducted for the rating procedure. This rating procedure may be expressed as:

$$SEER = \frac{\sum_{j=1}^{k} n_j \, X(T_j) \, Q_{ss}(T_j)}{\sum_{j=1}^{k} n_j \, \frac{X(T_j) \, E_{ss}(T_j)}{PLF(X)}}$$

where: SEER is the cooling seasonal performance factor which is the new figure of merit accounting for cyclic as well as steady-state effects.

$Q_{ss}(T_j)$ is the unit's steady state capacity at each bin temperature, T_j.

$X(T_j)$ is the load factor which is equal to the ratio of building load, $CBL(T_j)$, to the steady-state capacity, $Q_{ss}(T_j)$, when the steady-state capacity is greater than the building load. Otherwise it is defined as equal to one. It is approximately the percentage of the compressor on-time for each bin.

$PLF(X)$ is the part-load factor, which is a function of the load factor. More explicitly, $PLF = 1 - C_D\ (1 - X(T_j))$ where C_D is the cyclic degradation coefficient defined by the slope of a line on a normalized capacity (Qcyc/Qss) to normalized efficiency (EERcyc/EERss) graph which has been defined by the steady state full load test; and the cyclic test results.

$E_{ss}(T_j)$ is the steady-state power input for the particular outdoor temperature T_j.

Although the number of terms in Eq. (2) make the expression somewhat complicated, it is still of the same basic form as the inverse of Eq. (1). The numerator is the seasonal output or building load that must be met, and the denominator is the seasonal input which includes the penalty factor for cycling effects.

The value of C_D is assumed to be constant when in fact it is a variable. Complete characterization of the C_D variation over the entire load range would require unreasonable amount of testing. The cyclic test specifications given above should result in the best known single valued representation of the C_D variation. The rating procedures offer an option to the cyclic testing by accepting an assigned value of .25 for C_D. Based on measurements made on a variety of models presently in production, it would appear that different designs can have an average C_D value anywhere from .01 to .35. The optional assigned value (.25) is not intended to be a median or a goal of any sort. It was selected so as to encourage the manufacturer to compete in the marketplace by designing (and testing to verify) a more seasonally efficient unit, (i.e. one whose $C_D < .25$). On the other hand, those manufacturers who find the additional testing too burdensome still have a way to avoid increasing their testing costs without too drastic a performance penalty. Thus the 0.25 value was selected as a compromise between these opposing political-economic forces rather than for purely technical reasons.

For a heat pump having a single-speed compressor, it is possible to employ a simplified method to evaluate its SEER for an average U.S. climate. It can be shown that multiplying the COP obtained in the wet-coil test at 27.8°C (82°F) by the PLF evaluated at X = 0.5 (i.e. PLF = 1 - 0.5 C_D) yields a result which is virtually identical to the value of SEER obtained by applying Eq. (2) to national average bin data. This is equivalent to saying that the SEER can be found by evaluating the heat pump's dynamic COP at the national average cooling season temperature of 27.8°C (82°F) and the load factor (X = 0.5) corresponding to this average temperature.

Testing procedures for heat pumps in the heating mode require two certified test points taken under steady-state conditions at 8.3°C (47°F) and -8.3°C (17°F) outdoor temperatures and 21.1°C (70°F) DB/15.6°C (60°F) maximum WB indoor conditions. In order to establish a part load performance curve, it is necessary to require two additional test points to account for the cyclic effect and the frosting effect. The cyclic test point is reached directly from one test since the indoor coil is dry in the heating mode. The outdoor temperature value of 8.3°C (47°F) is prescribed as a matter of convenience since the traditional steady-state point (still used for capacity rating) is measured at this condition. This point does tend to be a good upper bound point, since typically the capacity curve will flatten at warmer temperatures. The frosting-point test is at 1.7°C (35°F). This is the point at which the maximum rate of frost might be expected to occur. A lower temperature condition would have less water vapor in the air, while a higher temperature condition might result in natural melting during the off-cycle. The frost buildup/defrost effect begins with the steady-state -8.3°C (17°F) point where no frost is assumed to occur, and causes the performance curves to deviate from the existing steady-state values in a linear direction through the 1.7°C (35°F) test point until 7.2°C (45°F), where a step change out of the frost region is assumed to occur. For convenience, this step change is defined at an edge of a temperature bin, and assuming that most outdoor coils have a 5.6 to 8.3 degrees C (10 to 15 degrees F) temperature difference between the air and the refrigerant, the 7.2°C (45°F) value seems to be reasonably representative of field behavior. The cyclic effect is superimposed on the frosting effect between the balance point and 7.2°C (45°F). At 7.2°C (45°F) and above, the heat pump's performance is degraded only by the part-load cycling effect.

The rating procedure is, as before, a matter of considering the cyclic capacity, cyclic power, and number of operating hours for each temperature bin and determining the weighted average for the heating seasonal performance factor. Algebraically, this may be expressed as:

$$HSPF = \frac{\sum_{j=1}^{k} n_j \, HBL\,(T_j)}{\sum_{j=1}^{k} \left[\frac{X\,(T_j)}{PLF(X)} \, \delta\,(T_j)\, E_{ss}\,(T_j) + RH\,(T_j) \right]} \tag{3}$$

where: HSPF, n_j, $X(T_j)$, PLF(X), $E_{SS}(T_j)$ have the same definitions as those in Eq. (2) except they are now applied to the heating mode.

$HBL(T_j)$ is the building heating load requirement which is shown in Fig. 1 and is defined by a zero value at 18°C (65°F) and the design heating requirement (DHR) value at the outdoor winter design temperature (T_{OD}).

$\delta(T_j)$ is the heat pump low temperature cut-out factor to account for those systems which have compressor that shuts off at a given outdoor temperature. It has a value of 0 if T_j < the cut-off temperature , a value of 1 if T_j > the cut-on temperature, and a value of 1/2 if T_j is in between these two temperatures.

T_j = 67 - 5_j is the representative temperature within the jth bin.

$RH(T_j)$ is the quantity of energy for resistance (supplemental) heat required for each bin. As illustrated in Fig. 1, it has a zero value at temperatures above the balance point and a finite value in the shaded triangular region below the balance point. It is, of course, quite sensitive to the sizing criteria and affects the HSPF significantly.

Both Eq. (2) and (3) are applicable to single-speed compressor/fan units only. The concepts discussed for these single-speed procedures are equally applicable for two-speed units, but require additional testing and more complex expression for the rating to account for the differences in operation at both speeds. Details are described in documents listed in TABLE B-3.

ANTICIPATED PROCEDURAL CHANGES FOR 1988

Although there was a multitude of technical issues addressed throughout the 1987 Rule Making proposed by DoE, the most significant was the inclusion of two systems never before addressed, the variable speed heat pump and the mixed-matched air conditioner. The following discussion is based on the NBS proposal to DoE after review of the Industry comments of the Proposed Rule Making.

The distinguishing feature of a variable speed system is that it compressor can operate at different speeds and allows the system to modulate its capacity within a certain range. Because of a capacity range it can provide, a variable speed system has two balance points in a given installation, and three modes of operation as shown for the cooling mode on Fig. 3. The low speed balance point, depicted in Fig. 3 as t_1, is the outdoor temperature at which the capacity line at the minimum compressor speed intersects the building load line. The high speed balance point, depicted in Fig. 3 as t_2, is the outdoor temperature at which the capacity line at the maximum compressor speed intersects the building load line. These two balance points separate three outdoor temperature ranges corresponding to three cases of operation of a variable speed unit. Operation of a variable speed system at the outdoor temperature ranges from 65°F to t_1 (case I) and from t_2 to 105°F (case III) is identical to operation of the two speed unit and appropriate portions of a two speed unit rating procedure can be applied to represent performance of variable speed equipment. However, for the outdoor temperature range between the two balance points (case II) no existing DoE (U.S. Department of Energy) procedure can represent adequately performance of a variable speed system.

The determination of the minimum (k=1) and maximum (k=z), speeds lines are done by steady state tests at the temperature indicated in Fig. 3. There is a cyclic test as well at 67°F (19.4°C) but it is felt that it will seldom be used since the default C_D = .25 results in a rather minor penalty. Where as it may theoretically amount to a 12.5% decrease in the SEER for a constant speed unit, it cannot exceed 5% for a variable speed unit with a capacity modulation ratio of 2 and less than 2.5% decrease for a unit with a modulation ratio of 3.

In order to use the same rating bin approach as indicated by equation (3) it is necessary to have a profile of the power input for the capacity profile that matches the building load line, which is fixed by the zero load point at 65°F (18.3°C) and a point approximately 91% of $Q_{ss}^{k=2}(95)$, (accounting for 10% oversizing). Empirical evidence indicates that this profile is parabolic in nature with the power profile deviating from linearity more strongly than either capacity or EER. Also, in order to minimize the amount of testing only one intermediate test point is necessary such that when combined with the balance points' power values a parabola can be fitted for the entire intermediate speed range. The intermediate speed test (k=i) is not straightforward since one has to decide at which speed to manually control the unit for the 87°F (30.6°C) test. To match the building load at the temperature is likely to be a lengthy hunt and seekd process in the laboratory. Therefore, the 87°F (30.6°C) test is specified to be run at 1/3 of the way between minimum and maximum speeds and the results are extrapolated to the building load line along an assumed constant speed profile whose slope is determined as the weighted mean of the maximum and minimum speed profiles. It is this crossover point, which will typically be very close to 87°F (30.6°C), of the capacity and power plots that is used to calculate the intermediate EER value $EER_{ss}^{k=i}$. The parabolic fit of the EER profile of the intermediate range in Fig. 4 and the linear building line of the same range in Fig. 3 are then used to determine the power values for each temperature bin by the equation: $P(T_j) = CBL(T_j)/EER(T_j)$

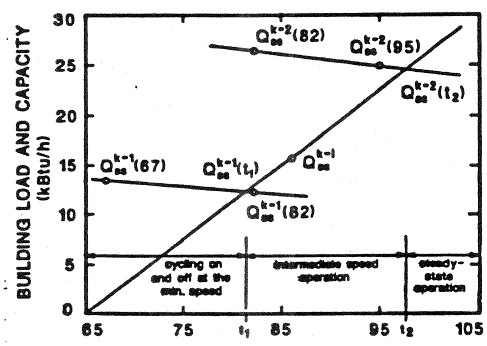

Fig. 3. Capacity of variable speed heat heat pump - cooling

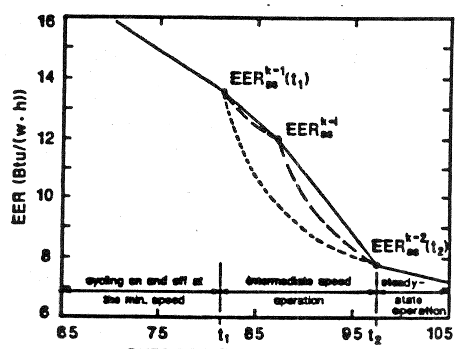

Fig. 4. Efficiency of variable speed heat pump - cooling

The interrelation between the building load and capacity of a variable speed system in the heating mode is shown graphically in Fig.5. The line originating at the 65°F temperature mark on the abscissa is the building load line. The set of two lines defined by system capacities at the maximum compressor speed, $Q^{k=2}(17)$, $Q^{k=2}_{def}(35)$ and $Q^{k=2}(47)$ provide a simplified representation of system capacity at the maximum compressor speed at different outdoor temperatures. Similarly, capacities $Q^{k=1}(47)$ and $Q^{k=1}(62)$ prescribe the system capacity line at the minimum compressor speed. An additional test point on the figure is $Q^{k=i}(35)$, system capacity at the intermediate speed test. The compressor speed during this test is to be the same as during the cooling intermediate speed test. Thus, the capacity may or may not fall on the building load as indicated in Fig.5.

The rating procedure prescribes a range of design heating requirements (DHR) in the heating mode for which the manufacturer has to evaluate the Heating Seasonal Performance Factor (HSPF) in different climatic regions. Although all HSPF values are available to a consumer, the HSPF value determined for a minimum DHR in Region IV has been selected for relative comparison of performance of all heat pumps for advertising purposes.

The building heating load line is prescribed by the equation:

$$BLH(T_j) = \frac{65 - T_j}{65 - T_{OD}} C * DHR$$

where: DHR = f (Q (47))
TOD = outdoor design temperature
C = 0.77, an experience based correction factor

It was found during development of the single speed equipment rating procedure that this prescription of the building load preserves a similar slope for the building load lines in both the cooling and heating. This slope similarity would not be maintained for a variable speed system if its maximum speed in heating is greater than maximum speed in cooling, and capacity at maximum speed in heating was used for DHR calculation. In such a case, the building load line would be unduly steep and the procedure would not provide the system with the rating credit for reducing the seasonal contribution of the electric heater. To alleviate this rating problem, the procedure prescribes a nominal capacity test in the heating mode at an outdoor temperature of 47°F. This test is applicable only to systems designed to run at a higher compressor speed in heating than in cooling and is to be performed at the maximum compressor speed used in the cooling mode. This test is optional and its result is to be used for evaluation of DHR only. If not performed, system capacity at 47°F at the maximum compressor speed has to be used for load calculations. The procedure for evaluating the intermediate power input is identical to that of the cooling mode.

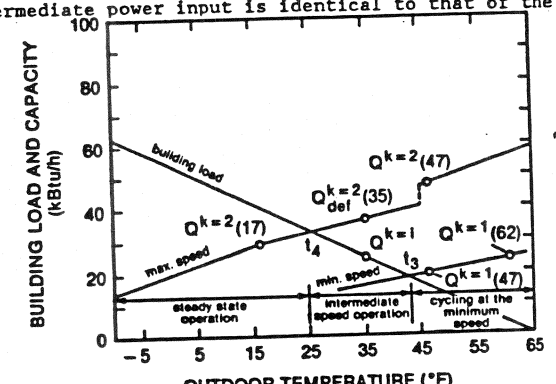

Fig. 5. Capacity of a variable speed heat pump - heating.

The need for the development of a mixed-matched air conditioner rating procedure arose primarily from the fact that there exists several "coil-only" manufacturers whose products are combined in the field with condenser units of a different manufacturer. This practice necessitated a procedure by which a condenser test result with one coil(matched, m) and a bench test coil (mixed, x) result could be combined to predict the performance of the condenser with the mixed coil. As a result of considerable laboratory data and system simulation studies the following relationships were developed for capacity:

$$Q_x + P_x = (Q_m + P_m)\, F_c^{.37} \cdot F_e^{\alpha} \tag{4}$$

and efficiency:

$$SEER_x = SEER_m \left(\frac{Q_x}{Q_m}\right)_{82} \left(\frac{P_m}{P_x}\right)_{82} F_{TXV} \tag{5}$$

The mixed system capacity, Q_x, is calculated by equation (4) using published rating data of the matched system's capacity, Q_m, and indoor fan power, P_m, the indoor coil scaling factor, F_c, and the expansion device scaling factor, F_e. The scaling factors are the mixed to matched coil capacity ratio and expansion device refrigerant mass flow ratios. The coil capacity ratio can be predicted by computer simulation if the simulation has been verified by the test of coils in the same family as those being simulated. Circuitry and coil air flow angle must also be accountable by the simulation. The expansion device flow factor and α coefficients are specified by NBS (see Table B-3 No. 6) for the various combination of devices (i.e. orifice, cap. tube, TXV) possible. This flow ratio is significant not only because it impacts the energy performance of the mixed system but also because it can affect the reliability of the compressor. For this reason, the range of F_e is restricted. The mixed system efficiency expression, $SEER_x$, equation (5), contains only one new adjustment parameter, F_{TXV}. The capacity and fan power data for the mixed system Q_x and P_x are based on similar test data as that in equation (4) except it is from the DoE test at 82°F (27.8°C) outdoor temperature instead of 95°F (35°C) as in equation (4). The matched data Q_m and P_m is required for 82°F (27.8°C) and that information is generally proprietary. Therefore, a rather complex expression for these two ratios $\left(\frac{Q_x}{Q_m}\right)_{82}$ and $\left(\frac{P_x}{P_m}\right)_{82}$, has been derived based on an assumed capacity increase of 5% when going from 95°F (35°C) and 82°F (27.8°C). The expressions used to determine the ratios (not shown in this paper) contain no other new data requirements. The F_{TXV} is a thermostatic expansion valve factor, of the order of 5% or less, used to account for the fact that the different (i.e. matched and mixed) expansion devices may have different off cycle bleed capabilities. For example, if the matched system had a capillary tube and the mixed, a non bleed TXV, the $F_{TXV} = 1.05$. This would suggest that the $SEER_x$ should be 5% better than the $SEER_m$ (all other things being equal) because the performance dynamics of the system would be improved due to the stoppage of the refrigerant draining into the evaporator during the off cycle (i.e. the C_D would be reduced).

This mixed-matched procedure of putting separate test results together to predict a system performance requires assumptions in a minor way about how a generic system performs. Therefore, it will always be somewhat less accurate than a test of "matched" system. It is therefore only recommended when the burden of test is to great to do otherwise.

FUTURE PROCEDURE DEVELOPMENTS

The future of test procedure development will obviously depend on the future developments in residential heat pumps and air conditioners. The law requires the Department of Energy (and thus NBS) to restudy the current evaluation procedure at least every five years to see if changes are necessary. Work, in another NBS Group responsible for water heaters, has already begun developing rating procedures for systems that simultaneously condition the space and heat the domestic water. Such systems already exist as desuperheaters and several companies have complete heat pump and water heater systems under development. Determining the load patterns for the various possible modes of operation would appear to be one of the more significant challenges for this system's rating procedure development.

The next five year period would seem to merit a consideration for change in the basic philosophy of not dealing with the control system during laboratory tests. Currently the units are manually controlled for all tests which are conducted under steady-state operating conditions. However, the potential operating complexities possible under a variable speed system with computer chip control of proprietary logic suggest that a more sophisticated test procedure may be merited. Such ideas as providing humidity control during the period that the space does not require cooling is now within the realm of heat pump control system possibility. Varying the fans speeds or coil area usage for energy performance improvement is possible. However, if one is to avoid placing an undue test and test facilities burden on manufacturers, the continuing challenge of maintaining a minimal test requirement as the basis for rating all possible operating modes must be met through innovative evaluation procedure developments. To this extent it is indented to give some consideration to the use of emulators as part of the test unit's operational control scheme during the laboratory evaluation period. It is hoped that emulator usage would allow for the heat pump system to be responsive to a prescribed load pattern to assist in determining all the possible operating modes the system could have. Once that is established, a series of steady state test evaluations could be developed that would characterize that system's performance overall. Dynamic testing is always a possibility but the state-of-the-art of some of the instrumentation transducers currently used would have to be advanced.

The balance of test burden versus accuracy of consumer information will continue to be a delicate one. But as systems grow more complex the problem is aggravated by the need of understanding how a generic system performs so that default factors or performance factors can be used to alleviate the ever increasing testing requirements. In the past the factors developed were possible because much was already understood about the reed valve, piston compressor. Similar public knowledge about scroll compressor performance and any other new component developments will certainly have to have a well reviewed history in the open literature if future procedure developments are to be done in a timely fashion. To this extent, the intra-industry/government cooperation that seems one of the keys to the Japanese industry's amazing progress should act as a model for the international industry as a whole so that the consumer world-wide can be better served.

APPENDIX A

The Department of Energy sampling requirements are designed to minimize the testing burden while maximizing the assurance that the test results of a few units may be applied to the entire product line. Specifically the requirements are that the true mean is not less than 95% of the adjusted sample mean.

For example, a manufacturer might test two units from a product line whose SEER rated values were 7.6 and 7.9. The mean value of these units is:

$$\bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i = \frac{1}{2}(7.6 + 7.9) = 7.75$$

with a standard deviation of:

$$\sigma = \left[\frac{1}{n-1}\sum_{i-1}^{n}(x_i - \bar{x})^2\right]^{1/2} = \left[\frac{1}{1}((7.6 - 7.75)^2 + (7.9 - 7.75)^2)\right]^{1/2} = .21$$

the lower confidence limit of the true mean is then:

$$\bar{x}_\ell = \bar{x} - \frac{t\sigma}{\sqrt{n}} = 7.75 - \frac{3.078\ (2.1)}{\sqrt{2}} = 7.29$$

where t = 3.078 is obtained from a table of one-sided percentiles of the t (Student's) distribution for 90% confidence and n-1 degrees of freedom. Finally an adjustment can be made to allow for the fact that the repeatability of the heat pump system test is considered no better than 5%:

$$\bar{x}_s = \frac{\bar{x}_\ell}{.95} = \frac{7.29}{.95} = 7.67$$

which is the SEER value this manufacturer can claim for this product line.

APPENDIX B

There exist many types of residential heat pumps in the U.S.
The formal documents that describe these testing and rating procedures are listed in Table B-1, B-2, and B-3.

TABLE B-1 Current ASHRAE Heat Pump Test Standards

1. ANSI/ASHRAE 37-78: "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment"
2. ANSI/ASHRAE 116-83: "Methods of Testing for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps"

The American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), has developed and maintains the laboratory test procedures relating to heat pumps (TABLE 1). Standard 37-78 is the document that outlines the steady state procedure, tolerances and test apparatus for all unitary units. One of the recommended test setups is illustrated in Fig. B-1. The concept is to measure the airflow and the wet bulb and dry bulb temperature differences across the indoor coil to determine the enthalpy difference. This difference is then confirmed within 5% by a similar determination on the refrigerant side for an acceptable capacity test. Electric power input measurements allow for completion of efficiency determinations. ASHRAE standard 116-83 has similar tests but also includes a series of cycling and frosting tests along with a temperature bin rating procedure for determining seasonal performance. Standard 116 is a result of a joint government/industry study after several years of experience with the Department of Energy (DoE) procedures.

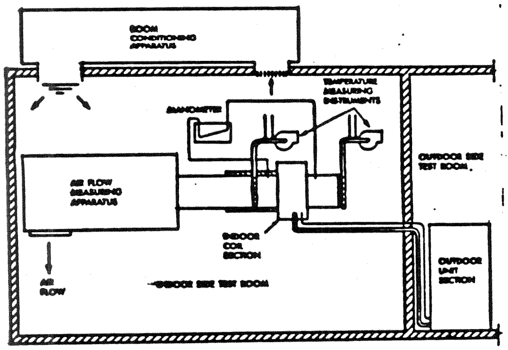

Fig. B-1. Tunnel air-enthalpy test method arrangement.

The DoE test procedures are identified by numbers 1 and 3 in TABLE B-3. As is the custom in this industry the ASHRAE test procedures do not include specific criteria (e.g. test operating conditions) that would allow for a numerical performance rating to be determined for a given test unit. These criteria were developed by the manufacturers through their industry association, the Air Conditioning and Refrigeration Institute (ARI), in conjunction with other interested parties. For the case of energy advertising and labeling the official NBS/DoE procedures for both testing and rating are NBSIR's 77-1271 and 80-2002.

TABLE B-2 Current ARI Heat Pump Standards

1. ARI 210-81[4]: Standard for Unitary Air Conditioning Equipment
2. ARI 240-81[4]: Standard for Unitary Air Source Heat Pump Equipment
3. ARI 210/240-84: Standard for Unitary Air Conditioning and Air Source Heat Pump Equipment
4. ARI 320-86[4]: Standard for Water Source Heat Pumps
5. ARI 325-86[4]: Standard for Ground-Water Source Heat Pumps
6. ARI 340-86: Standard for Commercial and Industrial Unitary Heat Pump Equipment
7. ARI 380-87[4]: Standard for Packaged Terminal Heat Pumps

TABLE B-3 NBS Test and Rating Procedures

1. NBSIR 77-1271[5]: "Method of Testing, Rating and Estimating the Seasonal Performance of Cental Air Conditioners and Heat Pumps Operating in the Cooling Mode"
2. NBSIR 79-1911: "Procedures for Testing, Rating, and Estimating the Seasonal Performance of Engine-Driven Heat Pumps"
3. NBSIR 80-2002[5]: "Method of Testing, Rating and Estimating the Heating Seasonal Performance of Heat Pumps"
4. NBSIR 80-2090: "Estimating the Heating Seasonal Operating Costs of Residential Hybrid Heat Pumps Systems, Including Units Retrofitted to Oil, Gas, and Electric Furnaces"
5. NBSIR 81-2434[6]: "Method of Testing, Rating and Estimating the Seasonal Performance of Ground-Water Source Heat Pumps"
6. NBSIR 86-3301[6]: "Rating Procedure for Mixed Air Source Unitary Air Conditioners and Heat Pumps Operating in the Cooling Mode"
7. NBSIR 88- [6]: "Recommended Procedure For Rating and Testing of Variable Speed Air Source Unitary Air Conditioners and Heat Pumps"

For an air-to-air residential heat pump the current official ARI standards are 210-81 and 240-81; these are used in conjunction with ASHRAE 37-1978. They are compatible with the industry standards in every way except they are more comprehensive, in that they require dynamic testing and seasonal ratings. In turn, ASHRAE 116-1983 and ARI 210/240-84 are slated to be adopted by DoE later this year as the new "improved" testing and rating procedures which will place industry and government requirements on an identical track. The DoE rating procedures are required for all heat pumps with performance ratings of single phase electric input and cooling capacity of 65,000 Btuh (19.45 kW) or less. Those procedures are mandatory (as they are for central air conditioners) as the basis of performance claims in any sort of efficiency advertising. This category will probably be

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[4] Basis for an AIR Certification Program
[5] Officially adopted by DoE and mandated for energy labeling purposes.
[6] Currently in the DoE rulemaking process.

expanded this year to include two new products: variable speed heat pumps and multizone heat pumps. It probably will not include water source, ground water, ground coil, solar assisted, or hybrid heat pumps. Finally, this year's DoE rulemaking will include rating mixed systems for the cooling mode at least. These are units that are a composite of the indoor and outdoor units, usually of different manufacturers. According to the NBS recommended methodology one coil in each family will be tested and the other coils' performance, may be predicted by a simulation which has been "tuned to" or verified by test.

NBS has defined coils which belong to the same family as those having in common all basic design features which in a prominent way affect heat exchanger performance. The important features that would place coils in different families are:

- basic configuration (A-shape coils, V-shape coils, slanted coils, flat-top coils, etc.)
- heat transfer surfaces on refrigerant side and air side (flat tubes vs. grooved tubes, different fin shapes on the air side),
- tube and fin materials,
- method of refrigerant distribution between coil circuits

One coil family may cover different coil sizes.

Mixed matched units have been a particular point of controversy in utility rebate programs and in various Federal qualification programs because of a lack of a mandatory rating methodology. ARI has developed a certification program but as with all their programs it is a voluntary program and not all coil manufacturers participate. It is anticipated that the DoE mandate will recognize ARI certification and the proposed NBS methodology as an acceptable rating method for mixed matched heat pumps in the cooling mode and eventually in the heating mode, as well. It is also likely that individual manufacturers will be allowed to submit their own methodology for approval.

In addition to the residential units under 65,000 Btuh that have been the focus of DoE attention, ARI considers units up to 135,000 Btuh (39.55 kW) cooling capacity including single and three phase motors (items 1, 2, 3, in Table B-2). They also have rating programs for water source (intrabuilding heat recovery) and packaged terminal (motel room type) heat pumps; also a generic group for commercial and industrial applications which include the largest unitary units made (greater than 500 MBtu/h or 145 kW). All of these types of equipment are evaluated in accordance with ASHRAE test standard 37-1978 and their respective ARI rating document (items 4, 5, 7 of Table B-2) by the manufacturer. For those product lines which are entered under an ARI certification program specific units are selected by ARI for test confirmation at ETL Laboratories in Cortland, NY, and capacity and steady state efficiency (i.e. EER or COP) results of the entire manufacturer's line are published semi-annually in a catalogue for public review.

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Over the past decade that my Group has been responsible for the development of the heat pump test and rating procedures I have been fortunate to have several very competent engineers to conduct the necessary research in the laboratory and on the computer. Most significantly was the initial project leader, Dr. George Kelly, (now Leader of our Division's Control's Group), who was largely responsible for the original documents published in the late 1970's, and current project leader, Dr. Piotr Domanski, who has been largely responsible for the current (1987) changes for variable speed and mixed-matched systems. Also contributing in many very important technical details was Dr. Walter Parken (deceased) and Mr. William Mulroy. Equally important from a program management and financial perspective has been Mr. Michael McCabe of the U.S. Department of Energy, under who's guidance this program continues to be directed.